DIESEL-ELECTRIC GENERATORS

The diesel, or compression-ignition, engine is one of the four principal types of internal combustion engine; that is, it is a machine that converts the chemical energy released from the burning of a fuel in an internal combustion chamber directly to mechanical energy. Although the diesel is a reciprocating machine, its mechanical energy is transferred from the engine by means of a rotating shaft that may be used to drive other mechanical, hydraulic, pneumatic, or electrical machines and equipment. Worldwide there are many diesel engines manufacturers, and the engine types available range from extremely powerful low-speed two-stroke engines of up to 70 MW to high-speed automotive-type engines to low-power portable units of less than 2 kW (1.492 hp). In industrial and marine applications, diesel engines are used mainly in the generation of electrical power, both ac and dc. In this article the topics addressed are the diesel engine itself and the production of ac power by diesel-powered generators.

GENERATOR APPLICATIONS

The main uses of diesel-generators are:

1. For base-load duties in locations where there is no utility supply—that is, usually in remote locations, on islands, or on ships and submarines.
2. As independent power sources where it is essential to ensure that a continuous supply of electrical power of acceptable quality is maintained at all times. Such systems are usually referred to as *uninterruptable power systems* or *no break systems.*
3. For "peak-lopping" or "peak-shaving" duties to limit the maximum or peak demand from a utility supply and so reduce the premium unit charge rate and hence the overall cost of the supply.
4. As standby or emergency power generation in case of major failure (*blackouts*) or partial shutdowns (*brownouts*) of the main or utility supply. (Such units are common in telecommunication centers, hospitals, mainframe centers, major financial institutions, and government buildings.)
5. Transportable (usually trailer mounted) generation units for providing temporary increases in the main supply especially in remote areas.
6. As part of a cogeneration, sometimes titled CHP (combined heat and power), plant.

Advantages of Diesel Engine Use

The main advantages of using diesel driven electrical power generators are (not in rank order):

1. *Performance.* Diesel engines normally have high thermal efficiencies, in the region of 40% and higher, almost

J. Webster (ed.), *Wiley Encyclopedia of Electrical and Electronics Engineering*.

regardless of their size. Some current state-of-the-art engines can achieve efficiencies over 50%, and engine manufacturers have forecast efficiencies as high as 60% by the twenty-first century.

2. *Maintenance.* Diesels represent mature and well-developed technology and are comparatively easy to maintain on site without the need for fully skilled personnel except for certain nonroutine tasks.
3. *Durability and Reliability.* Diesels have long lifetimes in the range, on average, of at least 20 to 25 years, and they can operate 7000 to 8000 h per year and in some cases up to 12,000 h between regular major overhauls.
4. *Fuel Efficiency.* In most power-generation applications, diesels have the most competitive fuel consumption rates, and between half-load and full-load their fuel consumption rate is reasonably constant. Depending upon the application, size of engine, loading, and the operating environment, diesel engines normally have a specific fuel consumption in the range 160 to 360 g/kWh. The new Sulzer Diesel RTA two-stroke engines are claimed to be able to produce up to 35,431 kW (47,520 bhp) with a specific fuel consumption as low as 154 g/kWh (115 g/bhp).
5. *Transportability.* Diesel-generators can be transported on purpose-built trucks or in specially equipped containers by land, sea, or air so that they can be used immediately on arriving on-site even in remote areas. For their physical weight and size, they can generate large amounts of electrical energy, sufficient to supply a small town.
6. *Cost.* The cost per unit power installed is very competitive, but it must be emphasized that in costing diesel-power generation it is crucial to determine the total installed costs, not simply the capital cost of the engine and the generator. As a general rule of thumb, the speed of crankshaft rotation basically determines the weight, size, and cost of an engine in relation to its output power.
7. *Operational Flexibility.* Diesels can use a wide variety of fuel quality and can be designed to use both liquid and gaseous fuels; that is, they are "dual–fuel" engines. They can also be adopted for use in cogeneration and total-energy systems and in "non-air" environments.
8. *Environmentally Compliant.* Diesels inherently produce low amounts of harmful exhaust emissions. However, in recent years, engines have had to be redesigned and exhaust-emissions treatment systems upgraded to meet increasingly stringent regulations. It is certain that further advances in the efficacy of emission reduction techniques will be required for all fossil-fuel power systems in the future.

ENGINE SIZE, CLASSIFICATION, AND SELECTION

Diesel engines are mass-produced with power ratings from 2 kW to 70 MW, and individual diesel-generator sets are available in unit sizes from 1 kVA to at least 55 MVA. Theoretically, individual diesel-generator sets could be grouped together to produce a power station of almost any output, but with few exceptions even the largest modern primary-power generation installations are limited, for most practical purposes, to between 150 and 200 MVA. The typical range and size of diesel engines and diesel-generator sets summarized in Table 1 are being constantly expanded, and compendiums such as the annual *Diesel and Gas Turbine Worldwide Catalog* (see Bibliography) should be consulted for the most contemporary information. A growing number of engine and generator manufacturers also now have extensive Internet web-sites.

Table 1. Engine and Generator Size Ranges

Application	Power Range (One Engine/Unit), kW	Speed Range (Maximum/Rated), Rev/Min
Automobile	35–100	4,000–5,000
Bus or truck	90–350	2,000–3,000
Stationary	1–150	1,200–3,600
	150–3,000	1,200–1,800
	100–15,000	300–1,200
Diesel generators	13–60,000	60–3,600
Dual fuel	93–1,900	375–2,600
Large marine 2-stroke	to 70,000	55–200

The most significant power range for diesel-generating plants, whether for continuous base-load, peaking, emergency, or standby electrical generation, is between 200 kW and 3.5 MW unit sizes. Diesel engines are not, however, classified by power output but are instead classified by shaft speed and are manufactured with maximum speeds of between 0.917 revolutions per second (rev/s) or 55 revolutions per minute (rpm, rev/min) to about 83.3 rev/s or 5000 rpm. This speed range is categorized into three commonly accepted classes:

High or higher speed:	Over 16.7 rev/s or 1000 rpm
Medium speed:	Between 6.67 rev/s and 16.7 rev/s, or 400 and 1000 rpm
Low or lower speed:	Under 6.67 rev/s or 400 rpm

The choice between low-, medium-, or high-speed engines will depend upon the application. However, when the diesel engine is used to drive an ac generator (alternator) and the crankshaft is directly coupled to the alternator's rotor, the shaft speed at which the engine must operate is dictated by the desired supply frequency, normally 50 to 60 Hz, and the number of alternator pole pairs; that is, n(rev/s) = frequency (in Hz)/number of pole pairs. Thus, for alternators with one to three pole pairs the crankshaft speeds will be 16.7 rev/s and 60 rev/s depending upon the supply frequency. All these speeds are indicative of the high-speed class of diesel engine. However, diesel engines which produce outputs in the hundreds of kilowatts and megawatt range tend to operate at much lower speeds, generally in the medium-speed category but sometimes in the low-speed class. When medium- and low-speed engines are used with alternators, the number of pole pairs is usually increased and there may be 70 or more for large-power machines in the MVA range.

Thus, in the selection of a diesel engine both power and speed are important, specifically the power produced at the desired continuous speed. Therefore, engine manufacturers specify their engines with a *rating*—that is, the power produced at a given speed under prescribed test conditions. At present the International Standard for the rating of *Reciprocating Internal Combustion Engines* is ISO 3046. However, for a given supply frequency, the choice of a particular engine and diesel-generator will depend upon and be influenced by many factors in addition to the engine rating.

Selection Criteria

In selecting a diesel engine for generator duty or a particular manufacturer's diesel-generator set, the following factors must be considered:

1. The fuel economy of the diesel at the desired maximum continuous speed.
2. The expected annual duty (hours of operation) of the generator.
3. The primary reason for the use of the generator: baseload, peak-shaving, no-break, standby, or emergency.
4. Initial capital cost including installation costs.
5. Continuing annual operating costs, including depreciation and loan interest payments (if any).
6. The security and reliability of the generated supply.
7. The supply quality—electrical.

Using these general criteria and other criteria pertinent to the specific application, such as currency exchange rates, local fuel and lubricating oil taxes, fuel quality availability, electrical supply and transmission regulations, environmental laws, availability of routine and emergency spares, and local technical skill level, it is usually possible to perform a comprehensive evaluation of the various levels of output quality and security of the generated supply and the associated global operational costs.

ENGINE AND GENERATOR SET RATINGS

An engine "rating" in terms of power and speed is specified by the manufacturer in accordance with an established "standard" which takes account of the purpose for which the engine is to be used. The standard specifies the test conditions, including fuel type and length of test, which must be employed to demonstrate the satisfactory performance of the engine. If the engine is to operate in surrounding conditions which differ from those prescribed in the standard, then it is necessary to adjust the rating to compensate for the differences. The standard will usually specify how to do this compensation. The rated power will be specified not only for particular test conditions but also for specific operational modes (e.g., *continuous* power related to a defined operating interval and *maximum continuous rating*) for marine applications. Rail traction rating and automotive rating are other self-explanatory application definitions.

In world markets, there are several established national standards by which ratings can be defined. Thus, when comparing the technical merits of engines manufacturered in different countries it is essential that account be taken of the differing test conditions under which the rating was established. To combat this problem, efforts have been made by a large group of countries to produce a common standard. These efforts have been coordinated by the International Standardization Organization in Geneva, and the International Standard, ISO 3046, previously mentioned is now used by many countries. Other countries and organizations have brought their own standard into line with the ISO while retaining their own title; for example, the British Standard BS 5514 is in full accord with ISO 3046. In the modern globally competitive world, the major manufacturers now also give details of engine ratings using the different standards. The most commonly encountered standards used by diesel engine and diesel-generator manufacturers are the German Deutsches Institut für Normung e. V., (DIN) series, the International Standardization Organization (ISO) series, the British Standards Institute (BS) series, and the US Society of Automotive Engineers (SAE) series, in particular DIN 6271, 70020, BS 5514, SAE J 1349, and, of course, ISO 3046. Organizations such as the SAE have a comprehensive standards service which can provide a wide range of national and international formation. The ISO 3046 specified test conditions are:

Ambient air pressure:	1.01325 bar
Maximum altitude above sea level:	0 m
Air inlet temperature:	27°C
Charge air coolant temperature:	27°C or 77°C
Fuel (lower calorific value):	42 MJ/kg

Unlike engines, diesel-generator sets have no rating systems, but there is a well-established terminology given in the various standards literature—for example, DIN 6280, part 4. However, as always, when selecting and purchasing any goods the "fine print" should be read carefully.

BASIC CONCEPTS AND WORKING CYCLES OF ENGINE OPERATION

The diesel engine is a compression-ignition (CI) piston engine which can operate with a variety of liquid fuels, or gaseous fuels in the dual-fuel mode, or with "solid" fuels such as pulverized coal dust. The fuel of choice is invariably liquid, but whatever the chosen fuel the principle of operation is basically the same. Air is induced into the engine and compressed by piston action to a pressure level an order of magnitude higher than at entry. As the pressure increases, so does the temperature, and when it reaches a sufficient value the fuel is injected into this air charge and the mixture spontaneously ignites. The pressure and temperature rises in the cylinder will be strongly influenced by the engine's *compression ratio,* which is defined as the ratio of the total volume of the cylinder at the bottom of the piston's stroke divided by the cylinder volume remaining when the piston is at the top of the stroke. Diesel engines usually have compression ratios in the range of 14 : 1 to 24 : 1, whereas spark-ignition (SI) engines, apart from high-performance versions such as used in motor sports, have compression ratios of about 9 : 1 in North America and slightly higher in Europe and some other countries. In an SI engine the fuel and air are induced as a mixture into the engine cylinder, and together they cannot be compressed to as high a level as air alone (as in CI engines) without the problem of "detonation" or "pinking" occurring. Thus, the temperature reached toward the end of the up-stroke in an SI engine is normally not sufficient to ignite the mixture, and some form of external ignition source is required—usually a spark plug.

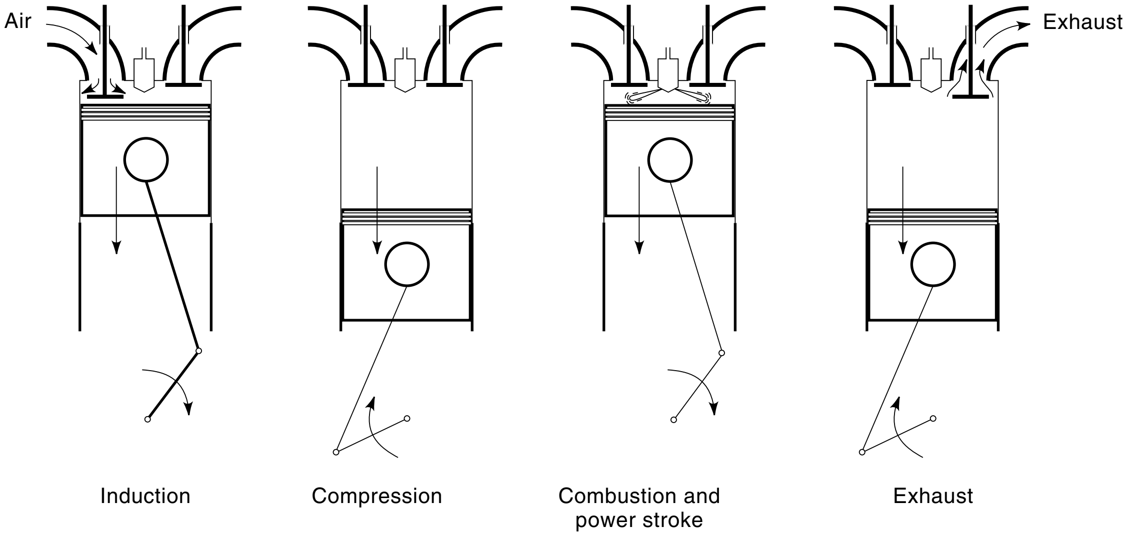

Figure 1. Four-stroke cycle.

As the compression ratio is increased, the hot gases expand more and more power is produced. Because diesel engines have an inherently higher compression ratio, then they should—and usually do—produce more power for a given quantity of fuel than SI engines. However, because the diesels use larger compression ratios (i.e., higher pressures), the engine components and block construction has to be structurally stronger, and invariably this means that diesels are heavier than comparable SI engines. For example, a 5.9 L (359 cubic inch) B Series turbo-diesel Dodge pickup engine has a mass of 440 kg (970 lb), whereas that of a Dodge 6.2 L (380 cubic inch) V-8 Magnum SI engine is only 245 kg (540 lb).

To achieve the combustion of the fuel–air mixture and the consequent production of power, CI and SI engines can use either a “two-stroke” or “four-stroke” cycle of operations, as now illustrated.

Ideal Four-Stroke

The basic four-stroke working cycle is shown diagrammatically in Fig. 1. The events take place in a particular sequence or cycle, hence the description.

1. *Induction Stroke.* The inlet valve is open, and the downward travel of the piston draws a charge of air into the cylinder. At the start of the stroke the piston is at its upper position in the cylinder, called top dead center (tdc). At the end of the stroke the piston is at its lowest position in the cylinder, called bottom dead center (bdc).
2. *Compression Stroke.* All the valves are closed, and the air trapped in the cylinder is compressed by the upward travel of the piston to a final pressure usually between 3.5 and 5 MPa (508 psi to 725 psi). The air temperature will rise to about 700°C to 800°C (1292°F to 1452°F). If the engine is supercharged, then the final pressures and temperatures will be higher. Near the end of this second stroke, a metered amount of liquid fuel will be injected by means of a pressure jet atomizer into the cylinder at pressures which, depending upon engine type and size, will range from 2 MPa (290 psi) to 150 MPa (21,756 psi). At the completion of this stroke the piston will be at its tdc position.
3. *Working Stroke.* All the valves are still closed. After a short time period between the start of fuel injection and the start of combustion, called the *ignition delay,* the fuel automatically ignites and almost completely burns. (In reality the fuel injection starts before the piston reaches the tdc position.) The temperature and hence the pressure further increase to as high as 2500°C (4532°F) and 7 MPa (1015 psi) and thus the piston is forced downward to its bdc position, thereby transforming the energy released in the combustion process into mechanical energy.
4. *Exhaust Stroke.* The exhaust valve opens and the piston moves up toward tdc, again driving the burned gases through the open valve.

The cycle then begins again. Because it takes two revolutions to complete the process, the valve gear and fuel injection equipment have to be driven by mechanisms that operate at half engine speed. Some of the power generated in the working stroke is stored in a flywheel to provide energy for the piston movements of the other three strokes.

Ideal Two-Stroke

In a two-stroke engine (Fig. 2) the need for separate induction and exhaust strokes is eliminated by combining (1) the working and exhaust strokes and (2) the induction and compression strokes. The crankcase is sealed and serves as an air storage reservoir, and the mechanical operated inlet and exhaust valves in the main cylinder are replaced by ports.

1. *The Intake-Compression Stroke.* The piston travels to its tdc position and, in doing so, physically blocks or closes the transfer port and then the exhaust port. Thus, the trapped air charge is compressed. Simultaneously the underside of the piston is drawing a fresh charge into the crankcase through a nonreturn inlet

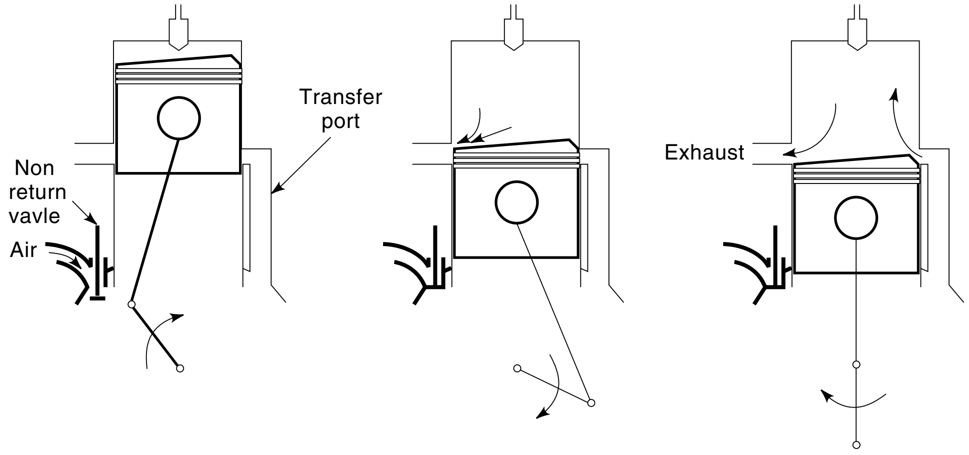

Figure 2. Two-stroke cycle.

valve. As the piston approaches tdc, fuel is injected into the cylinder.

2. *The Power-Exhaust Stroke.* After the ignition delay period the fuel–air mixture ignites and combustion takes place, causing the gas temperature and pressure in the cylinder to rise. This forces the piston to travel toward its bdc position. In this movement the exhaust port is uncovered, allowing burned gases to vent; the process is called *blowdown.* When the piston reaches bdc the transfer port is uncovered and the air charge in the crankcase that has been compressed—by the piston motion—expands and passes to the top of the cylinder. The cycle then repeats.

Two-Stroke Versus Four-Stroke

In terms of the number of manufactured units, most diesel engines are four-stroke. However, two-stroke diesel engines are popular in marine applications where their inherent operational characteristics can be used to advantage. These marine two-stroke engines are usually low-speed high-power machines, and when used in alternator applications a large number of pole pairs is usually required. A comprehensive discussion of the relative economic, thermodynamic, and technical merits of two-stroke and four-stroke engines is beyond the scope of this article, but the following points are worth noting.

1. In the two-stroke engine the “power” stroke occurs in each crankshaft revolution, whereas in the four-stroke engine it occurs once in every two revolutions. Thus, a two-stroke engine of the same mass and volume as a four-stroke engine will produce more power. However, for many reasons the two-stroke will not produce twice the power, and in some instances the two-stroke’s power advantage is only marginal.
2. At the same rotational speed, four-stroke engines are more fuel-efficient than two-strokes. There is less time with a two-stroke to induce the air charge and exhaust the combustion products, and this often leads to the fuel supplied not being completely burned. However, over the past decade there have been major improvements in the fuel efficiency of low-speed two-stroke engines, and the majority of large marine two-stroke engines have specific fuel consumption rates (in the 160 g/kWh to 190 g/kWh range) which are lower than those in most medium- and high-speed four-stroke engines.
3. Four-stroke engines require mechanically operated valves but two-stroke engines do not; constructionally, this enables the two-stroke to be simpler. However, the more important attraction of the lack of valves is that two-stroke engines can run in either direction, clockwise or counterclockwise, and therefore do not require a reversing transmission. This is a particular advantage in marine propulsion applications.
4. In a two-stroke the residence time of the fresh air in the engine is lower, and therefore there is less time for this air to cool the cylinder. Thus engine overheating is more likely to occur with a two-stroke engine, and the design of the cooling system needs more attention than does that of a four-stroke engine.

PRACTICAL ASPECTS OF ENGINE OPERATION

There are a number of practical impediments which prevent the exact idealized two-stroke and four-stroke cycle of operations, previously described, from being realized in practice. These barriers are generally associated with the capability of the engine to burn all the supplied fuel effectively over the full range of desired operating conditions. The level of capability is strongly influenced by combustion chamber and fuel injection system design and by the *gas-exchange processes* which govern the flows of air into, and the combustion products out of, the engine. The actual operation of a diesel engine can also be purposely modified by using supercharging, exhaust gas recirculation, or synthetic atmosphere operation.

Scavenging and Valve-Timing

It is of paramount importance that all the burnt gases be expelled. Residual gas left in the cylinder will reduce the amount of fresh air that can be induced; and if sufficient air is not available, then incomplete combustion will usually be the result. In a two-stroke engine it is very difficult to expel all the combustion products. As a consequence, the power stroke produces far less thrust than in the ideal case. To improve the gas flows in a two-stroke the engine is designed so that when the intake and exhaust ports are cleared by the

Table 2. Heisler's Survey of Valve Timings

Inlet valve opening before tdc	10°–30°
Inlet valve closing after bdc	40°–75°
Exhaust valve opening before bdc	40°–75°
Exhaust valve closing after tdc	10°–30°

piston, some mixing of fresh air charge and burned gases is encouraged, the idea being to use some of the incoming air to help expel the exhaust gases. This process of clearing the combustion products is called *scavenging.* The majority of engines use "loop scavenging," so-called because of the pattern of the flow mechanism, but in marine engines a "uniflow scavenge" technique may be encountered. Supercharging two-stroke diesel engines also aids the scavenging process. In the four-stroke engine, however, nearly all the burned gases are forced out of the combustion chamber, allowing an almost complete air charge to be drawn into the cylinder. The power stroke in a four-stroke engine therefore produces relatively more power than its two-cycle counterpart, somewhat offsetting the advantage of the latter's one power stroke per revolution.

The superior gas-exchange processes in present four-stroke engines are partly the result of ensuring that the timing of the opening and closing of the inlet and exhaust valves are controlled rigorously. In the idealized case, either both valves are closed or one is open and the other closed and vice versa, but this is not so in practical engines. The closing of the exhaust valve is usually delayed, whereas the opening of the inlet valve is usually advanced. This has benefits in the "emptying and filling" processes of the cylinder, especially at higher speeds, but the imposed "valve-overlap" can lead to the engine producing higher emission levels because of incomplete combustion. This disadvantage is more frequently encountered with four-stroke SI engines than with diesel engines, because the latter use unthrottled air or inhale only fresh air. In supercharged diesel engines, changes in valve timing can be used to offset any detrimental valve-overlap problems while maintaining the advantages of such control. Heisler (1) carried out an extensive survey of valve timings for various engine types. These timings are summarized in Table 2.

Combustion Systems

In simple terms the amount of fuel which can be combusted in a given time will strongly influence the amount of power produced, whereas the completeness of combustion will dictate the engine's efficiency and hence its fuel economy. The level and type of emissions produced are also governed by the combustion quality. While compression–ignition engines are generally more efficient than their prime mover competitors, much depends upon the size of engine, the design of the combustion, and the fuel injection systems.

When liquid fuel is injected into the engine, it is in the form of a fine spray of droplets which vaporize and mix with the already present air forming small nuclei from which combustion is initiated. These nuclei spread rapidly, ensuring complete combustion of the fuel. The mixing of the fuel and air is crucial to the operation of the engine. Ideally the combustion chemistry only requires the so-called *stoichiometric* amount of air to be present per unit fuel (for hydrocarbon fuels the stoichiometric air–fuel ratio is between 14 and 15 parts to 1 by mass). However, at present it is not practically possible to design an engine that will burn all the fuel at the stoichiometric ratio, and the amount of air supplied must exceed this ratio to ensure complete combustion. This is not a problem with the diesel, unlike the SI engine, because the fuel and air are fed separately to the cylinder, allowing air–fuel ratios to be made much higher. At idle speeds, a diesel's air–fuel ratio can be as high as 85 : 1, or even 100 : 1, whereas at full load this ratio will be between 25 : 1 and 30 : 1. Thus, diesels always have *weak* air–fuel mixtures.

The shape and configuration of the combustion chamber have a major impact on the efficiency of the fuel–air mixing process. Thus, not surprisingly, a great deal of attention has been paid by engine manufacturers to the design of diesel combustion systems, and this has resulted in the development of a variety of systems which can be conveniently classified into two main groups: (1) indirect injection (IDI) Systems and (2) direct injection (DI) Systems. Engines using the former type are sometimes called *prechamber* or *divided-chamber* engines. These secondary titles are literal descriptions of the engine's combustion system. In both IDI and DI engines the main combustion chamber is formed between the suitably shaped moving piston crown and the stationary cylinder head and to a much lesser extent the cylinder walls.

In an IDI engine, fuel is injected into a hot prechamber or a swirl chamber in the cylinder head from which leads a narrow passage to the main combustion chamber. In the former case, fuel is injected, at a relatively low pressure of up to 30 MPa, into the prechamber which has been heated by the exhaust gases. Combustion starts and the partially combusted gases are rapidly driven into the main chamber, thereby generating intensive mixing. In the swirl-chamber configuration a ball-shaped or disk-shaped auxiliary chamber in the head is connected to the main chamber through a tangential throat that is slightly wider than the passage used with prechamber designs. A strong air vortex is generated in the swirl chamber during the up-stroke of the piston. Fuel is injected into this swirling air, and the design of the chamber ensures that the mixture initially ignites in a hot wall zone. As the combustion starts, the air–fuel mixture is forced into the main chamber where it mixes with the remaining combustion air.

In DI systems, which are the type most frequently encountered in stationary power applications, the fuel is injected directly into the combustion chamber above the piston. The cylinder head is usually flat and the geometry of the combustion chamber is largely dictated by the clearance volume between the piston and the cylinder and the form of the physical depression in the piston crown. Many different depression forms are available, and numerous others have been tested, but the cylindrical piston recess is the most commonly encountered in modern engines. Because the injection, evaporation, and ignition process must take place much more quickly in the DI engine, the intake valve ports are designed to produce vortices during the induction and compression strokes in a similar manner to the IDI swirl-chambers. The direct-injection principle is also universally employed in modern medium-speed and on many high-speed engines.

IDI engines are especially popular in Europe because of their superior ability to handle variable fuel qualities and produce lower levels of pollutants than DI engines. Their maximum speed can usually be higher than DI engines; hence the IDI engine, especially when used with a turbo-charger, is most suitable for passenger car use. The use of prechambers and swirl chambers allows the air–fuel ratios in IDI engines to be lower than in normal DI engines. Thus for the same power output the IDI engine will be physically smaller than the DI equivalent. However, engines with small cylinder diameters tend to have lower thermal efficiencies because their high surface area-to-cylinder volume ratios give larger heat losses. Thus, small indirect injection engines may have maximum thermal efficiencies below 30%, whereas DI engines of comparable power but which are physically larger can have efficiencies as much as 20% higher. In most cases, for the normally encountered compression ratios, the minimum useful cylinder volume for a diesel engine is about 0.4 liters, although smaller engine capacities are available.

The DI engine in terms of fuel economy is generally superior to the IDI version, but its fuel-injection system tends to be far more complex. Developments aimed at reducing emissions for DI engines are reportedly meeting with more success than those associated with improving IDI fuel economy. However, for applications where physical space is at a premium and high shaft speeds are necessary in the 5 to 200 kW range, IDI engines are competitive especially when environmental compliance is an overriding factor.

Fuel Injection

As the amount of intake air remains constant at any particular engine operating point, it is only the fuel quantity which needs to be regulated. Thus, the function of the fuel injection system is to supply the engine with fuel in quantities exactly metered in proportion to the power required and timed with the utmost accuracy (usually to at least 1° of crankshaft angle) so that the engine will deliver that power within the limits prescribed for fuel consumption, exhaust smoke, noise, and gaseous emissions. In general terms the injection period and the pressure increase with the size of the engine. A complete diesel engine fuel system consists of a fuel tank, fuel filter, supply pump, an injection system, and a governor. The injection system includes an injection pump, delivery ducts, and injector nozzles. The pressure difference across the nozzle(s) orifice is very high, typically between 15 MPa and 160 MPa, to ensure that the fuel is atomized and the spray droplets on entry to the combustion chamber have sufficiently high penetration velocities to traverse the chamber in the time available and thus fully utilize the available oxygen (air). The nozzles are the most important part of any fuel-injection system, and they must be designed to very high standards, precision manufactured and scrupulously maintained.

In the early diesels, fuel was injected by a blast of highly compressed air, but this technique has long been overtaken by “airless” injection pumps which generate the required pressures. For the past 70 years the most widely used mechanical fuel injection systems have employed *jerk*-pumps, of which there are two principal types, namely, in-line or “camshaft-actuated” pumps and rotary or “distributor” pumps. Medium-speed and higher-speed engines tend to use individual in-line pumps for each cylinder. Higher-speed engines employ jerk-type pumps which are driven from a self-contained camshaft auxiliary drive from the engine. The large two-stroke engines often use a common rail system, wherein fuel is maintained at a constant pressure by the use of an accumulator system. Each cylinder has a fuel valve that is driven and timed from the main engine camshaft.

In multicylinder engines, to ensure that each cylinder operates in an identical manner, the periods of injection, the timing, and the delivered quantity must be precisely matched. Many fuel pumps incorporate an engine governor. On distributor pumps the governor is usually an integral part of the assembly where with in-line pumps the governor is a separate add-on unit. The governor–fuel-pump combination must give a straight-line relationship between load and engine speed. This requirement is especially important for diesel-generator applications because any deviation could reduce the effective run-up in a particular load range to below the stability limit.

The injector, the device that introduces the fuel by spray into the combustion space, is in effect a spring-loaded needle valve, whose tip covers the injector nozzle hole(s). The shape of the combustion chamber will largely dictate the design of the nozzles in terms of the number of holes, their angles, and the angles of spray. Thus, each unique type of combustion chamber design requires its own special nozzle. The throttling pintle nozzle is used on prechamber and swirl-chamber engines, whereas the hole-type nozzles are used with the nondivided chambers of DI engines. Because diesel fuel systems must have very small passages and nozzle holes, it is of paramount importance that the fuel supplied to the injection systems is well-filtered. Less than 5–10 g of hard abrasive particles in the 5–20 μm size range will wear out most commercial multicylinder pumps.

There are many different fuel injection systems available and a multitude of available design philosophies. The design, modeling, and manufacture of fuel injection systems is a very complex subject. Excellent starting points for such a study are the publications of Bosch (2), Heisler (1), and Stone (3).

Air Charge Pressurization and Inter-cooling

To produce more power from an engine of fixed size, it is necessary to burn more fuel in the combustion chamber. This can only be done by supplying more oxygen (i.e., air). In an engine drawing in normal ambient air (*naturally aspirated*) the fresh air charge will have a temperature and pressure at the end of the suction stroke essentially the same as the surrounding atmosphere. The density of this aspirated air regulates the mass of fuel which can be burned during the working stroke, and this in turn determines the maximum power that can be developed. If, however, the air supplied to the engine is pressurized at the same temperature, then the density is increased and for the same volume flow a greater mass of air (oxygen) can be fed to the engine and hence more fuel can be burned. This pressurization process is called *supercharging*.

A compressor is used to pressurize the intake air, and the two main methods of driving this compressor distinguish the form of supercharging used. If the compressor is driven from the engine crankshaft (usually by chains or gears), the system is called “mechanically driven supercharging” or, more commonly, “supercharging,” whereas if driven by a turbine powered by the engine’s exhaust gases the system is called “turbocharging.” In the early 1990s the Brown-Boveri company redeveloped their novel variant of the supercharger known as Comprex or the pressure-wave supercharger. However, the basic Comprex supercharger can only operate over a narrow performance speed and load range. The use of two-stage supercharging using a combination of a mechanical supercharger and a turbocharger is also starting to become more common.

Pressurizing the intake air can increase the power output by as much as 60% over an equivalent naturally aspirated engine of similar speed and dimensions. However, because the power required to drive a mechanical supercharger must be provided by the generated shaft power, some of the gains made by supercharging are lost. This is not the case with a turbocharger because the system is powered by energy that would otherwise be "wasted." As much as 30%, and maybe more, of the energy input to the engine may be rejected in the exhaust gases, and it is the recovery of this energy that makes the turbocharger method of supercharging particularly attractive.

However, turbochargers invariably suffer from "turbo-lag" following increases in engine load and speed. To match these changes, some of the energy in the exhaust has to be used to alter the speed of the turbocharger's rotor. The time taken for the new speed to be attained will depend upon the rotor's inertia and the efficiency of the turbocharger. It will take a turbocharger several seconds to respond to large step changes, 80% to 85%, in load demand. Provision also has to be made to prevent the turbocharger from overspeeding and therefore creating excessive in-cylinder pressures that could damage the engine and its components. This is achieved by the use of a "waste-gate" or valve in the exhaust system that enables some of the gases coming from the cylinder(s) to bypass the turbocharger.

Although the inlet charge density is increased by the rise in pressure across the compressor, it is partially decreased by the accompanying increase in temperature as the air is compressed. Higher engine inlet temperatures have the advantage of reducing the ignition delay period, but the thermal loading on the engine is increased and the full gains of supercharging are not realized. However, this loss is recoverable by the use of charge air coolers (intercoolers) placed downstream of the turboblower. The lower air intake temperature has the further effect of reducing not only the maximum cylinder pressure but also the exhaust temperature, and with it the engine's thermal loading. The increase in output power over an uncooled turbocharged engine can be as high as 25%, especially at higher engine speeds. Further improvements in fuel economy are also obtained by the use of intercooling, particularly at part loads below about 45% of rated power.

Gas-to-gas heat exchangers are bulky regardless of whether the air-pressurization system's intercooler is made an integral part of a radiator or is a separate unit upstream of the radiator. Alternative water-cooled arrangements use either a separate radiator circuit for charge-air-cooling water or, in marine installations, seawater-cooled heat exchangers with finned tubes carrying the seawater and over which the charge air passes. Even so, despite its advantages intercooling is not universally used. The provision of a secondary cooling system can be logistically difficult and the added cost and complexity cannot always be justified, especially for low-power applications.

Engine Governors

Almost without exception a diesel engine must use a device—a *governor*—to maintain a desired speed accurately. The required accuracies are specified in standards akin to the "rating" standards, and they are usually a subpart of the main diesel standard (e.g., ISO 3046, DIN 6280). Without a governor an engine at idle would either eventually stall or the engine speed would continue to increase until the engine self-destructed. A governor automatically controls the engine speed by regulating the fuel supply. There are two general classes of governor: (1) *isochronous,* where the engine speed is maintained at a set value, irrespective of changes in load, and (2) *variable speed* or *droop,* which has the facility to adjust the set speed of the governor. There are many different forms of these two governor types ranging from simple mechanical devices to sophisticated electronically controlled electromechanical and electrohydraulic servo-systems. The modern critical needs for fuel economy and environmental compliance have led to increasing sophisticated governor systems being developed.

With a diesel engine each crankshaft speed is associated with a given maximum torque that has been determined by testing. If the load on an engine is removed and the amount of fuel supplied stays the same, the speed will increase at a rate proportional to the load change. This phenomenon is termed "speed droop," and a major characteristic of a governor system is its speed droop. The measure of speed droop is normally referred to as the maximum full-load speed by a formula of the type

$$\delta = \frac{n_{\text{lo}} - n_{\text{vo}}}{n_{\text{vo}}} \times 100\% \qquad (1)$$

where δ is speed droop, n_{lo} is high-idle (maximum) speed, and n_{vo} is maximum full-load speed. A reasonably large speed-droop factor normally increases the stability of the complete engine system, but its value is usually limited by the operating conditions and application. Consequently, because of the usually demanded high quality of alternating current supplies the speed drop for diesel-generator sets is restricted to a maximum of 5% and often less. Governors used with generator sets almost always have other functions to perform in addition to speed control for normal synchronous speed operations. The selection of a governor for such purposes involves satisfying a number of criteria associated with (1) the application requirements (e.g., backup, emergency, or isolated loads) and (2) the prime mover requirements. The most common considerations are:

1. Steady-state speed control
2. Steady-state speed regulation
3. Transient response
4. Parallel and automatic operation

When two or more diesel-generator sets are run in parallel and proportionally share a variable total load the governor system must also be designed to ensure that each machine experiences (1) an equal drop in speed (regulation) from no-load to full-load (kW) and (2) an equal reduction in voltage from no-load to full-load kvA (reactive). Electronic governors are suited for such parallel generator operations because fully isochronous load sharing can be more readily achieved because of their faster response than mainly mechanical or hydraulic governors. Models of engine governor operation can be found in many electrical and control engineering textbooks and on a number of internet websites.

Fuels and Lubricants

The same standards organizations which provide guidelines for most aspects of diesel-generator operation also provide similar documentation for fuel and lubricants. In addition, other organizations such as the American Society of Mechanical Engineers (ASME), the American Petroleum Institute (API), the American Society for Testing and Materials (ASTM), and national defense agencies also provide specifications. In stating ratings, manufacturers will often state not only the calorific value of the fuel as implied by the rating system used but also state other fuel information. Density, viscosity, flash-point, and in particular the *cetane* number of diesel fuel are usually provided along with other properties. The latter is very important because it is a measure of a fuel's ignition quality. In general the higher the cetane number the better the ignition quality and the shorter the ignition delay. High-speed engines tend to use cetane numbers between 45 and 55; medium-speed, 35 and 45; and low-speed, 25 and 35. These ranges are very approximate. The cetane number is determined from engine tests but another measure, *diesel index,* is an indication of a fuel's quality based on certain physical characteristics of the liquid such as aniline point and API gravity. The correlation between cetane number and diesel index is not exact and is not always reliable. The sulfur, carbon residue, ash and sodium, and vanadium content of diesel fuels are all also important parameters because high values of any of these substances will normally lead to high wear rates and other maintenance problems. The standards organizations take all these factors into account as well as cetane number and produce recommended lists of fuel "classes" for particular applications and operating speeds.

With lubricating oils the viscosity is very important because it is a measure of its resistance to flow. A "thin" oil has a low viscosity, whereas a "thick" oil has a high viscosity. As the temperature of a single type of oil decreases, so does its viscosity. Oil types are classified or graded by their viscosity values as defined at specified low and high temperatures, and the system developed by the SAE is now almost universally accepted. The designators SAE 0W, 5W, 10W, 15W, 20W, and 25W are used to indicate oils in order of their low-temperature viscosities. SAE 20, 30, 40, and 50 are oil classifications in terms of viscosity values at 100°C, these are the thicker grade oils. Multigrade oils use designations such as SAE10W/20, and so on.

Although diesel engines are tolerant to fuels and lubricants of varying qualities, it is advisable, where possible, to use the recommended grades and to test the state of the lubricating oil at prescribed intervals. A full discussion on fuel and lubricants is not possible here, and for those who wish to pursue the topic the publication by Lilly (4) provides excellent coverage of the fundamentals of lubrication practice. Up-to-date information on fuel and lubrications can be obtained from the periodic SAE publication, *Fuels and Lubricants Manual.*

ENVIRONMENTAL ASPECTS

With the heightened societal concerns over the global environment and, in particular, the potential health and ecological risks associated with the emissions produced by the burning of fossil fuels, much more attention than ever before is now being paid to the "environmental performance" of power generation systems, electrical and mechanical. In the technical literature the term *environmental compliance* has been created to describe the environmental performance of an engine or system compared with the requirements of a specific set of regulations or codes of practice. In the United States, for gaseous emissions, the Clean Air Act is the legal basis for pollution control. Passed in 1967, the Act has been amended a number of times, the most recent major change being in 1990. Other countries have similar legislation, but not all. Emission limits are defined not only for engine type but also by application. Much of the legislation regarding fossil-fuel engines has been aimed at the use of such engines in automobiles and large public supply–sized power generation stations. However, in many countries the environmental laws are now being expanded to include a broader range of engine applications.

The exhaust from a diesel engine is a complex mixture of organic and inorganic compounds and gas-, liquid-, and solid-phase materials. The organic compounds largely originate from lubricating oil and unburned fuel and include classes of compounds such as aldehydes, alkanes, alkenes, and aromatic compounds. Of the latter the solid-phase polynuclear aromatic hydrocarbons (PAHs) have been identified as having an overall carcinogenic effect, especially when inhaled in a confined working environment. There are also health risk concerns about the total particulate matter (TPM) found in fossil fuel exhausts. Some 90% of particulates from diesel have sizes in the range 0.0075 μm to 1 μm, and these can be readily inhaled and trapped in the bronchial passages and alveoli of the lungs. Where it has been possible to precisely assess the risks and to produce consistent data on the effects, legislation or recommended operating procedures have usually followed. However, the human health significance of specific components of diesel exhaust is still a matter of intense debate.

The inorganic substances found in diesel exhaust include sulfur, oxygen, carbon, and nitrogen-containing compounds and oxides of nitrogen, NO_x. Some of these compounds originate in the fuel, especially sulfur and, obviously, carbon and sometimes in very small quantities, nitrogen. The main environmental focus has been on the generation levels of NO_x and carbon monoxide (CO), and increasingly stringent pollution laws now define the acceptable limits of particulates, NO_x, CO, unburned hydrocarbons (HCs), and smoke which may be generated from the burning of carbon-based fuels. (Diesel exhaust smoke is further categorized into white/blue and black/gray.) However, there is a multiplicity of regulations which define limits using different units—for example, parts per million, (ppm), g/mile, g/bhp-h, g/kW, $\mu g/m^3$, and so on. The limits are also application-dependent, and different values are defined for the same application depending upon which test is used to determine the emission levels. The situation is then somewhat confused.

Present environmental legislation still appears to be more concerned with the effects of pollution rather than its cause. In law, a substance has to be assessed as being toxic prior to deciding which measures have to be taken for its control. Fortunately, many engine manufacturers and fuel producers have taken preemptive measures to ensure that concerns over “suspect” substances are addressed. In other instances, emission level guidelines proposed by authoritative nonlegislative organizations are being adopted, but not in every instance.

Although great efforts are being made to produce more environmentally friendly diesel fuels and lubricants, better engine designs and emissions control equipment, these measures by themselves will not ensure perpetual environmental compliance throughout the operational life of an engine. Operational vigilance is most important, including planned maintenance, inspection and emission monitoring, and the enforcement of good working practices.

OPERATIONS WITH NON-AIR INTAKE MIXTURES

The term *non-air* originates in the description of the engine being non-air breathing—that is, operating in an environment where access to atmosphere air is prevented. The most obvious example of such an environment is underwater. In these situations an oxidizing atmosphere has to be produced which fulfills the same role as normal atmospheric air. Diesel engines operating under such circumstances may be called artificial or *synthetic atmosphere diesel* (SAD) engines. However, there are situations where the “non-air” breathed by the engine may be because of contamination (e.g., underground operations), or the air may be mixed with recirculated exhaust gases as part of an emissions reduction strategy. In the latter case the term *exhaust gas recirculation* (EGR) is used to describe the system. Although not uncommon, the application of the term “non-air” to such systems is debatable and can be confusing. A further definition problem can be encountered as SAD engines use exhaust gas recirculation in the formation of their synthetic atmospheres. Thus great care must be taken when reviewing the published technical data of “non-air” diesel engine operation.

Modified Air—Exhaust Gas Recirculation

Of the many techniques that have been developed to prevent the formation of NO_x, one of the most successful, especially with SI engines, is that of EGR. EGR is normally achieved by the use of an external system that returns a portion of the exhaust gases to the intake manifold or the combustion chamber. The basic concept is illustrated in Fig. 3. (Attempts have been and are being made to use valve timings control to allow exhaust gases to mix with intake air to produce an EGR effect; this is sometimes called *internal* EGR). The primary purpose of the use of EGR is to reduce the peak in-cyclinder temperature. This temperature has a threshold band of values below which, usually, NO_x will not form. Once the temperature reaches about 1700 K the amount of NO_x formed in the combustion process increases rapidly. The influence of EGR on the level of NO_x emissions is the result of (1) the increased heat capacity of the carbon dioxide contained in the recirculated gas, which produces a smaller temperature rise and thus a lower peak temperature, (2) the reduced partial pressure of oxygen resulting from the replacement of part of the combustion air with exhaust gas that contains a lower concentration of oxygen, and (3) the combined effect of (1) and (2), which reduces the combustion velocity and also leads to a decrease in the temperature rise. All these factors reduce the amount of NO_x formed and may, almost wholly, prevent its formation if the threshold temperature is not reached.

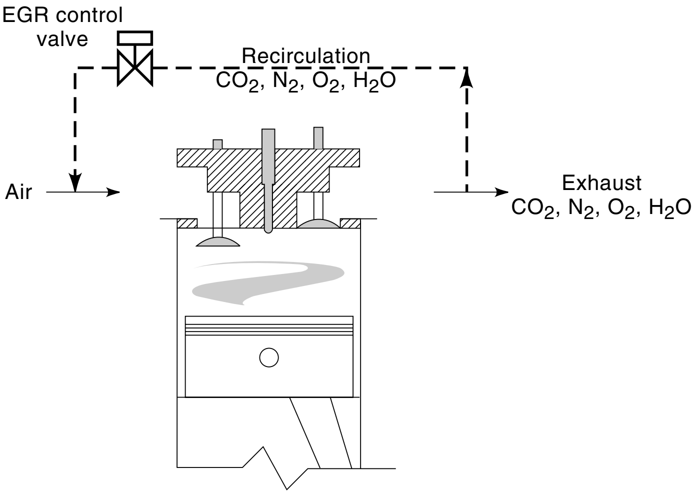

Figure 3. EGR basic concept.

In practice it has been found that EGR is more effective at part loads and that the amount of EGR used has to be limited. This is because the greater the rate of EGR, the less the amount of “fresh” air entering the engine system and hence the lower the air–fuel ratio. At some stage there will be insufficient air present to guarantee complete fuel combustion, and an increase in the emission levels of pollutants other than NO_x will occur. Thus, although the application of EGR technology can achieve very significant reductions in the levels of NO_x emissions, up to 50% at half-load, there can be some deleterious effects, especially at higher rates of EGR:

1. Particulates, smoke, unburned hydrocarbon, and carbon monoxide emissions may increase.
2. Under certain engine and EGR operating conditions, slight decreases in engine performance and fuel efficiency (~5%) can be experienced.
3. When used with turbo-diesel systems, high percentages of EGR at high engine load conditions can place excessive demands on a turbocharger if no intercooler is used. These effects are summarized on Fig. 4.

Mathematical Definition of EGR

The widely accepted SAE definition of EGR as *a system which returns a portion of the exhaust gases to the combustion chamber* covers both internal and external devices. Unfortunately, there are a multitude of mathematical definitions in current use, but all these can be summarized into the three basic formulas given as Eqs. (2) to (4). The numerical values obtained from the formulas are not equivalent because the relationships between the values are dependent upon the operating conditions prevailing in both the engine and the EGR system. Thus, great care must be used in the interpretation of EGR data.

Mass-Based EGR:

$$\%\mathrm{EGR_m} = \frac{m_{\mathrm{EGR}}}{m_{\mathrm{intake}}} \times 100\% = \frac{(1 - m_{\mathrm{air}})}{m_{\mathrm{intake}}} \times 100\% \qquad (2)$$

where m_{EGR} is the mass flow rate of recirculated gas, m_{intake} is the total inlet charge mass flow with EGR, and m_{air} is the mass flow rate on intake fresh air ($m_{\mathrm{intake}} = m_{\mathrm{EGR}} + m_{\mathrm{air}}$).

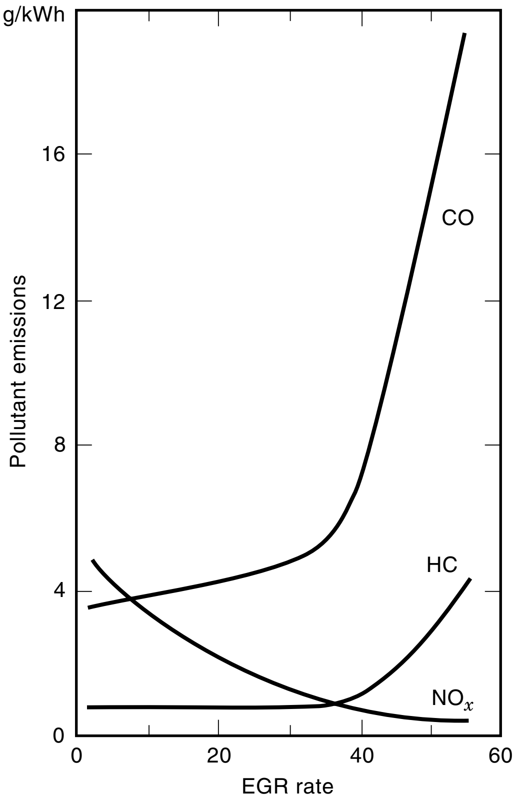

Figure 4. Effects of EGR (Bosch).

Volume-Based EGR:

$$\%\mathrm{EGR_v} = \frac{V_{\mathrm{aO}} - V_{\mathrm{aEGR}}}{V_{\mathrm{aO}}} \times 100\% = \frac{V_{\mathrm{EGR}}}{V_{\mathrm{aO}}} \times 100\% \qquad (3)$$

where V_{aO} is the volumetric flow rate of intake air without EGR, V_{aEGR} is the volumetric flow rate of intake air with EGR, and V_{EGR} is the volumetric flow rate of recirculated exhaust gases.

Carbon Dioxide Concentration-Based EGR:

$$\%\mathrm{EGR}_{\mathrm{CO_2}} = \frac{\%\mathrm{CO}_{2(\mathrm{intake})} - \%\mathrm{CO}_{2(\mathrm{amb})}}{\%\mathrm{CO}_{2(\mathrm{exh})} - \%\mathrm{CO}_{2(\mathrm{amb})}} \times 100\% \qquad (4)$$

where intake stands for intake concentration, exh stands for exhaust concentration, and amb stands for ambient concentration.

In actual operation most present EGR systems simply have an on–off valve which returns a metered amount of exhaust gas to the intake for a set time period. However, more sophisticated systems are becoming available which are part of an overall electronic management system for the engine or generator set. Even so, more research is required on the use of EGR especially with larger engines before the technology can be considered to be mature. For example, *cold* EGR in which the returned gas is cooled has been shown to offer advantages under certain operational conditions, but whether hot or cold EGR should be the standard technique used is still open to question.

A prime concern with the use of EGR systems is not only the increase in particulate formation but the fact that these solid abrasive particles could be fed to the cylinder. Ingestion of these solids significantly and rapidly increase the rates of wear, erosion, and corrosion of the engine parts and also degrade the quality and effectiveness of the lubrication oil. Therefore, it is very important that adequate filtration systems be used to prevent these solids from entering the working spaces. The NO$_x$-particulate problem is not unique to EGR systems because, in general, in-cylinder NO$_x$ reduction methodologies are plagued with particulate increases because the conditions that favor NO$_x$ reduction are normally detrimental to the oxidation of particulate matter and vice versa. Nevertheless, EGR appears to be one of the most promising techniques for NO$_x$ control.

Synthetic Atmosphere Systems and Operations

It was the need for better marine and submarine engines at the turn of the century that gave the diesel engine its first application successes. The diesel was used to provide safe and reliable power for the propulsion of a submarine when it was on the surface, which it was for most of the time. Underwater, secondary batteries were used to power electric motors to provide propulsion power and meet the "hotel" load—that is, all nonpropulsive power demands. The energy storage capacities of secondary batteries was and still is very limited. Thus, the underwater endurance of diesel-electric submarines was restricted to a few hours—and less if high speeds were used. Moreover, it was logistically inconvenient to have to use two separate power systems. Consequently, submarine power system designers started to explore the possibility of running diesel and other internal combustion engines underwater using stored oxidants. Several synthetic atmosphere or air-independent power (AIP) diesel engine designs were patented in the first decade of the twentieth century and ran successfully. The advent of the First World War and an unfortunate accident involving an underwater SI engine lead to a curtailment of SAD engine development. Subsequently the surface diesel/lead-acid secondary battery combination became, and still is, the standard *conventional* submarine propulsion system (Fig. 5). It should be noted that submarines which are not powered by a nuclear reactor are described as conventional.

Although interest in the SAD engine never subsided completely, it was not until the 1980s that commercial units became available. The main reason the concept was resurrected was the rapidly escalating effectiveness of electronic surveillance equipment which had made it necessary for even conventional submarines to stay underwater longer. Furthermore, the increasing exploration and exploitation of ocean

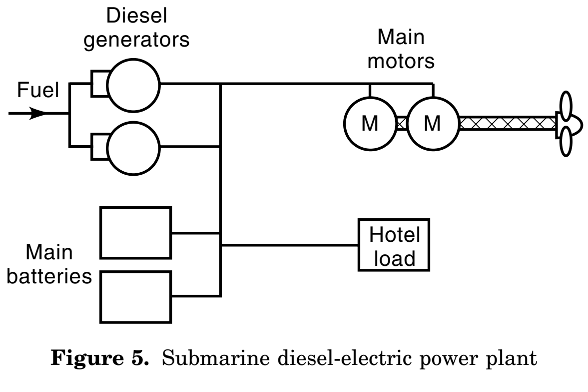

Figure 5. Submarine diesel-electric power plant

resources also led to the need for commercial underwater vehicles with better performances than could be obtained from existing electrochemical batteries. Today SAD engines are available in Europe for fitting or retrofitting in underwater vehicles of up to 2500 metric tons displacement.

In a submerged underwater vehicle, since the supply of air for combustion is limited, an artificial or synthetic atmosphere has to be created. The solution is, in principle, relatively simple: A mixture of pure oxygen is mixed with recirculated exhaust gas to create the desired atmosphere. However, since for combustion eight times more oxygen is required than fuel, the storage of oxygen becomes the dominant problem. Modern AIP heat engine systems usually store the oxygen in liquid form, but this requires an expensive and complex cryogenic plant. Compressed oxygen can be used with commercial submarines since their submerged endurance requirements are less demanding.

Another immutable problem is the disposal of the surplus exhaust products because not all the gases can be recirculated for the reasons discussed earlier when considering NO_x reducing EGR systems. These products, if vented to the surrounding seawater, have to be at a pressure above that prevailing at the vessel's operating depth. Normally this means that a gas compressor is required. At depths below about 200 m, the power requirements of the exhaust compressor are an unacceptable burden. A number of novel solutions have been proposed to this problem, but the most common one is to chemically or physically "scrub" the exhaust of all or part of its major constituent, namely, carbon dioxide. If all the carbon dioxide is removed, it is necessary to add another moderating gas to the intake "air" mixture such as argon or nitrogen. Such additions normally result in better engine performance but can be logistically inconvenient.

Another alternative is to use high levels of carbon dioxide with excess oxygen. However, as the carbon dioxide proportion is increased, it is necessary to preheat the mixture if ignition is to be guaranteed. If there is more than about 40% carbon dioxide in the intake mixture, the performance of the engine is significantly degraded. An energy-efficient compromise is to use a mixture of oxygen, added moderating gas, and 15% to 25% recycled carbon dioxide. The amount of exhaust gas which has to be expelled is thus reduced and the compressor's power needs are decreased, enabling overall system performance levels to be attained which are equivalent to those of naturally aspirated engines.

There are three basic forms of SAD engine: *recycle, closed cycle,* and *semiclosed cycle.* The main difference between the three systems is the method adopted for the management of the exhaust gases (Fig. 6).

1. *Recycle Diesel (RCD).* In the RCD mode, some of the exhaust gases are discharged from the system and the remainder are circulated. Carbon dioxide will be the main constituent of the recycled exhaust since it is usual to condense out the H_2O from the combustion products.
2. *Closed Cycle Diesel (CCD).* In the CCD system the combustion products are chemically scrubbed from the exhaust. The moderating fluid, normally nitrogen, is unaffected by the scrubbing process and is recirculated.
3. *Semiclosed Cycle Diesel (SCCD).* The SCCD is a combination of the RCD and CCD systems, where the exhaust gas is scrubbed with seawater. Alternative moderating gases, such as argon, can also be used with these systems to enhance engine and system operation.

With the current AIP diesel engine research development programs, cost-effective solutions to long-standing problems are being found by the use of "off-the-shelf" technology. However,

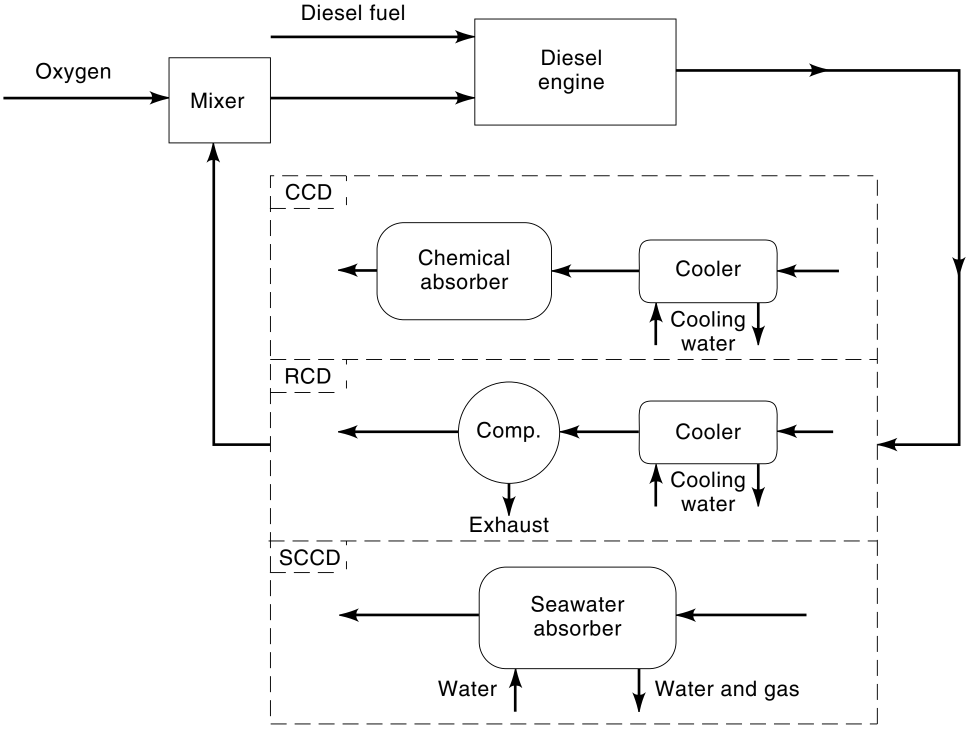

Figure 6. Synthetic atmosphere diesel.

underwater vehicle designers must continue to ensure that any proposed AIP diesel system, whether a "stand-alone" or "add-on" system, is assessed as part of an integrated vehicle design process. This will also involve the choice of generator because the main use of the present AIP systems is the underwater recharging of the secondary batteries. The designer can choose to remove some of these batteries and replace them with an AIP diesel system, or the vessel can be made larger to accommodate the extra system. The choice will depend upon the vehicle size and intended mission. As yet, the cost of AIP systems is more in harmony with their military use where mission effectiveness is more important than cost effectiveness.

AC GENERATORS

When used for generator applications the diesel engine is usually a multicylinder unit classified by the synchronous speed required, the type of fuel to be used, and the physical arrangement of the cylinders (e.g., in-line, Vee, and whether four-stroke or two-stroke). Although special engines are made for such purposes, they are more commonly variants of engines used for other stationary applications. As given in Table 1, there is a wide range of diesel-generators available including a growing market for dual-fueled and cogeneration sets. There are over 50 major manufacturers of complete diesel-generator sets and separate ac generator units for use with diesels.

Operating Principles and Construction

There are two basic ways in which an ac generator may be constructed: (1) a rotating armature with a stationary field or (2) a rotating field with a stationary armature. Almost all medium- and large-sized ac generators are of the latter form of construction. The reasons for this can be readily illustrated with a simple example. Suppose that a 30 MVA, 13.8 kV, three-phase synchronous generator is required. The line current would be $30 \times 10^6/1.73 \times 13{,}800$, that is, 1257 A. If the machine used stationary poles with a rotating three-phase winding, then three slip-rings would each have to handle 1257 A. The insulation of each ring and associated brushgear would be subjected to a working voltage of 13.8/1.73, that is 7.98 kV. However, a rotating field would only require two slip-rings, and these would have to handle the exciting current not the line current. Even if the power required for exciting the poles were very high (-250 kW at 400 V), the exciting current would be 625 A, less than half the line current, and the insulation would only have to handle 400 V. As lower currents and voltages have to be handled, the brushgear and insulation are not as massive and the construction is much simpler. This leads to a number of advantages which enable machines of this type of construction to handle high operating voltages (up to 33 kV), handle higher speeds, and produce a greater output than a similar-sized rotating armature machine.

Ac generators are also broadly classified into two groups by their rotor construction. These are *salient* or projecting pole machines and *cylindrical* rotor or nonsalient pole machines. With the former type, which invariably have a relatively large diameter but short axial length, the pole pieces—up to six—are usually dovetailed and keyed to the rotor shaft. When a greater number of pole pieces are required, these are bolted to the rotor. This latter method is less secure, but the machine will operate at a lower speed because of the increased number of pole pairs. The majority of diesel-generator sets use salient pole rotors, especially the larger units, and, in general, the larger the output, the lower the speed. However, a high-speed generator will cost less than its lower-speed equivalent.

An increasing number of modern diesel-generator sets are *brushless,* especially those of over 20 kVA. In these machines the excitation is obtained from an ac exciter via a three-phase bridge rectifier. The main exciter is a rotating aramature-type ac generator, and the dc field poles are bolted to a static magnetic ring that is part of the exciter's main frame. The armature conductors are arranged in a three-phase, star-connected distributed winding, permanently connected to the main generator field winding. With conventional machines the rotor windings are excited by dc exciters and slip rings with their associated brushgear. Such machines can react much quicker than brushless machines to load removals. Conventional excited machines are encountered, particularly in marine generator sets. However, although there have been significant improvements in the design and manufacture of conventional machines, they inherently require more maintenance than the brushless versions because of the problems caused by the presence of carbon and copper dust.

Additional Selection Factors. In selecting a diesel-generator set there are a number of technical factors to be considered in addition to those already listed earlier for the economic assessment of their use. The following are some of the more influential factors.

Voltage. Most countries have preferred ranges of output voltages which have specified regulation requirements and quality limits. In North America and Europe where 60 Hz supplies are the most commonly encountered, the voltages found in practice are 2.4, 4.2, 6.9, 13.2, 13.8, and 15 kV. The majority of plants generating over 5 MW use an operating voltage of 13.8 kV, whereas those in the 1 MW to 4 MW range usually operate between 4.2 and 6.9 kV. In the United Kingdom, the ranges for three-phase 50 Hz supplies are 415 V (to 1.5 MVA), 3.3 (0.5 to 6 MVA), 6.6 (0.8 to 10 MVA), and 11 kV (1 to 20 MVA). The present trend is for voltages to be increasing. Output voltages normally have to be held to $\pm 5\%$, which can be easily handled by modern automatic voltage regulators. In most industrial applications a voltage regulation of 40% to 45% at a 0.8 power factor will satisfy most needs.

Parallel Operation. As discussed earlier, when two or more diesel-generators are run in parallel it is important that they run in synchronism while proportionally sharing the variable load. The distribution of load between generators operating in parallel can be varied more efficiently and effectively by varying the driving torques of the engines rather than by varying the exciting currents. Consequently, active power-sharing is a function of the engine and governing system. The load division is achieved by changing the speed setting on the governors. Increasing the speed on one unit will result in an increase of load on that unit, and decreasing the speed will result in decreasing the load. High-quality speed sensing is therefore very important.

An alternative method of operation when using speed-sensor control is to have one engine as a master for isochronous operation and the other engine or engines in the plant as slaves for droop operation. The amount of droop, typically 5%, should be the same for each generator paralleled into the system. Electronic governors are becoming more popular, and these sense the electrical load on each generator rather than rely on speed measurements. This technique enables better transient response to load changes and allows more flexible mounting arrangements.

Startup. For emergency generating duties and no-brake installations it is important that the engine can start immediately and accept the load very rapidly without degrading the lifetime reliability of the engine. It is usual with such units to provide lubricating and coolant heaters so that the liquids can be kept at constant temperature of 27°C to 30°C. The use of a high cetane number fuel will assist the unaided cold-starting of an engine. However, special starting systems—electric, pneumatic and hydraulic—are produced by most diesel-generator set manufacturers for specific duties. For critical installations and for operations in very cold climates, startup aids include ignition-assisting glow-plugs and special starting mixture systems such as the Kold Ban (of Illinois, US) ether–air units.

Protection, Installation, Instrumentation. The construction of portable generator set containers and the installation housings of fixed site sets must meet laid-down standards, namely, BS, DIN, and so on. However, it is always essential that the adequacy of the ventilation system be ascertained. Filtration of the incoming and exiting air is very important, and in extremely dirty conditions it is advisable to use close-circuit ventilation. Temperature monitoring of the strategic engine and generator components is necessary to avoid damage to the set from overheating. Where ventilation is a concern the demands and duty on the set should be reduced.

Electrical protection devices will be needed to isolate short-circuit faults and clear them as soon as possible to avoid plant shut down. Voltages, currents, frequency, and power meters should all be available on generator sets/installations. For parallel operation a set of synchronizing instruments and controls are required. The advent and availability of microprocessor-based instrumentation, control, and monitoring systems has enabled physically small single-unit packages to be used for engine, generator, and supply condition monitoring.

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GRAHAM READER
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